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A Feasibility Study into Increasing the Rotational Speed of the Tuner in the DSTO Electromagnetic Reverberation Chamber

Frank Weeks and Grant Philp

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ABSTRACT

This note presents a feasibility study into reducing the test time in the Defence Science and Technology Organisation (DSTO) Electromagnetic Reverberation chamber by increasing the rotational speed of the tuner. The analyses cover the tuner's structural integrity, the drive motor capacity, and whether the drive motor software can be modified to accommodate the increased speed.

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Executive Summary

Data gathering time when using the DSTO Electromagnetic Reverberation chamber is a considerable proportion of the total test time. If the rotational speed of the tuner can be increased test times can be reduced. The analyses consider a tuner speed of 4rpm which will reduce testing time to one quarter of that presently taken.

Assuming that the paddle assembly acceleration and deceleration rates are maintained at their original values, the paddle assembly structure has been found to be sound at the new rotational speed of 4rpm. The increased stress in welds due to wind loading was found to be insignificant and had little effect on the original safety factors.

Drive motor capacity was found to be adequate, with the torque required to accelerate the paddle assembly being half the specified motor torque. With the original acceleration and deceleration rates being maintained, the torque loadings are unchanged.

The drive motor control software will have to be modified in order to maintain the original tuner acceleration and deceleration values, since the reduction ratios of the gearbox will have to be reduced. The necessary changes to the drive motor control software are feasible.

Contents

1. Introduction

The time taken to gather data when using the DSTO Electromagnetic Reverberation chamber is a considerable proportion of the total test time. If the rotational speed of the tuner can be increased, test times can be reduced.

For tests where a continuous tuner rotation is required, data gathering is limited to the period of one revolution. This paper addresses the feasibility of increasing the tuner speed from 1rpm to approximately 4rpm, thus reducing the data gathering time from 1 minute to 15 seconds.

The objectives of this note are to recommend a method for increasing the tuner's rotational speed, and to review the structural integrity of the energy tuner.

2. Method for increasing rotational speed

To increase the maximum rotational speed of the tuner assembly from 1rpm to 4rpm, the gearbox ratio will have to be changed.

The present motor to tuner speed reduction is accomplished by a sealed gearbox, with a reduction ratio 30.16:1, and two sets of toothed belt gears, each having a reduction ratio of 6.86:1, giving a total reduction of 1419:1.

The easiest and least expensive part of the gearbox system to change is the toothed belt gear section. By replacing the top 2 gears, the desired tuner speed of approximately 4rpm can be achieved.

2.1 Gearbox ratios

motor gearbox 30.16:1
lower toothed belt 6.86:1
upper toothed belt 6.86:1
Giving a total ratio 1419:1 (Tuner speed of approximately 1rpm)

With a motor speed of 1440
new ratio to give a tuner speed of 4rpm = 1440 ÷ 4 = 360:1

From above, ratio of motor gearbox and lower toothed belt gears = (30.16:1) x (6.86:1)
= 206.9:1
Therefore the ratio required for upper belt to achieve a new gearbox ratio of 360:1 = 1.74:1

From the manufacturer's catalogue [1], the nearest gear sizes to achieve this ratio is a 144 tooth gear and a 80 tooth gear, giving a gear ratio of 1.8:1, total gearbox ratio of 372.4:1 and tuner speed of 3.9rpm.

The present 192 tooth and 28 tooth gears can be replaced with the newly selected gears with a minimum of modifications to the gearbox. The present toothed belt can be used, but the belt-tensioning block will have to be modified. Photo 1 shows the present gearbox configuration and, Figure 1 shows the new gears, drive belt, and belt tensioner positions.

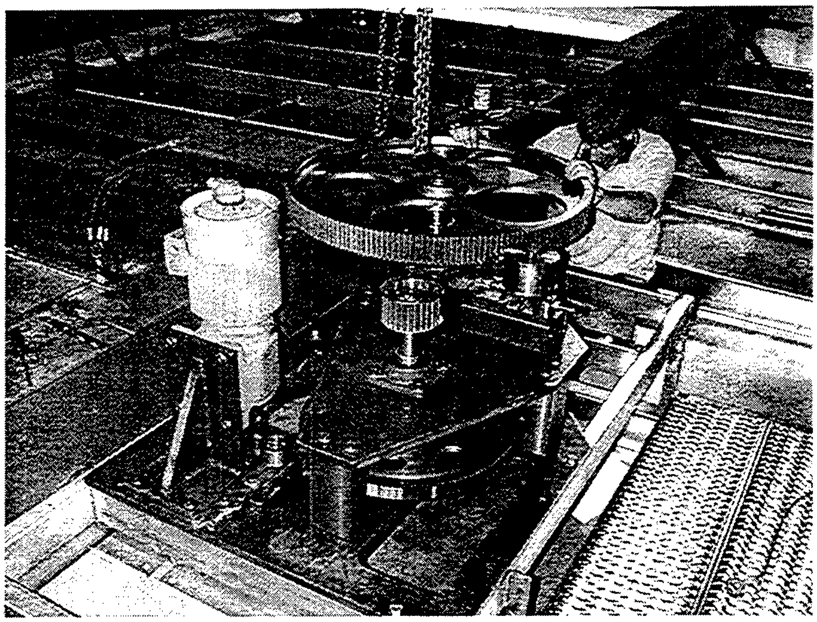

Photo 1 Gearbox assembly

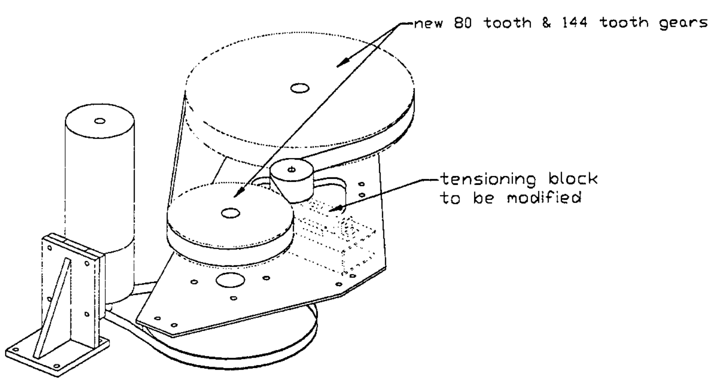

Figure 1 Gearbox assembly, showing replaced gears

3. Motor Controller

3.1 Drive motor

In the continuous running mode the electric drive motor runs at 40% of its rated full load current, and during tuner acceleration runs at 60% to 70% of its rated full load current.

It is proposed that the present tuner acceleration and deceleration rates be maintained, that is with the tuner now taking 4 times longer to accelerate to the required 4rpm. Therefore the motor current under acceleration will be the same as before the modifications.

During continuous rotation the drive motor load will be approximately the same as the present load. The increase in paddle wind resistance was calculated, and is considered negligible (see Appendix 1 for wind loading calculations). Since the motor is presently running at 40% of its full load current rating, it is anticipated that the increase in tuner speed will not significantly increase this current. It is recommended that the motor current be monitored during the tuner controller set to work period.

3.2 Motor controller software

Due to the change in gearbox ratio, the motor controller software will have to be modified to maintain the present tuner acceleration and deceleration rates. The motor controller software has been checked and it can be modified. The present software uses 46% of the motor controller's memory, leaving adequate room for software changes. The software is stored inside the motor controller on EEPROM. It is written in S.E.W.'s own language, which closely resembles Assembler. The software is developed and downloaded to the motor controller using P.C. based software communicating with the motor controller over a RS232 serial link.

3.2.1 Continuous rotation mode

It is considered that the motor controller software will be able to accommodate the changes for the continuous rotation mode. Motor acceleration and deceleration rates will have to change, with the maximum motor speed staying the same at 1440rpm. Acceleration ramps will be extended to maintain present tuner acceleration rates, it will now take 30 seconds to reach full speed of 4rpm. Software modifications are considered complex because of the requirement to override the motor controllers hardware acceleration ramps.

3.2.2 Stepping mode

The motor controller software for the tuner stepping will need to be completely rewritten. A new requirement is the ability to do a 5° backward step. The tuner velocity will need to be recalculated constantly to implement the 30 second acceleration ramps. The software will need to calculate the maximum velocity for the present step size, calculate deceleration points and do final positioning of the tuner. The software will need to be able to dynamically recalculate these values if the instrumentation controller issues additional stepping instructions. This section of software is considered complex to implement although portions will be common with the continuous rotation mode software.

4. Structural Integrity

Since acceleration and deceleration rates are unchanged whether the tuner assembly rotates at 1rpm or 4rpm, the original loads weld stresses, and safety factors, as calculated in [2], are correct.

4.1 Wind loading at 4rpm

Wind loading calculations can be found in Appendix 1 of this document.

When the tuner assembly operates in the continuous rotation mode, the side welds on the hub root absorb the torque due to wind loading. The torque due to wind loading was found to be very small at 12.7Nm per paddle and results in a safety factor of 13,790. This indicates that the wind loading has little bearing on the over-all stress at the paddle hub and can therefore be ignored. At 4RPM each paddle has a wind loading of 12.7Nm, Appendix 1, this is insignificant compared to the inertial loads of 1762Nm experienced during start up and stopping [2].

5. Motor torque during continuous running

Total torque due to wind load for all 4 paddles at 4 RPM = 50.6Nm from Appendix 1
New gearbox ratio = 372:1
Therefore torque at drive motor due to wind loading = 50.6 ÷ 372 = 0.136Nm
From SEW Eurodrive catalogue the maximum motor torque is 10.16Nm

The extra torque on the drive motor due to wind loading is 1.34% of the drive motor's torque rating. As the original current drawn by the drive motor is only 40% of its full

load rated current, it is concluded that the drive motor size is adequate to cope with the increased rotational speed of 4rpm.

6. Summary

To increase the tuner rotation speed to approximately 4rpm, the present 192 tooth and 28 tooth gears can be easily replaced with 144 tooth and 80 tooth gears respectively, achieving a gearbox ratio of 372.4:1. This modification can be incorporated with the minimal of change to the gearbox assembly.

6.1 Tuner structural integrity

The tuner is structurally sound if original acceleration and deceleration rates are maintained. The extra load due to wind resistance is so small that it can be ignored. The original stress calculations and safety factors are still valid.

6.2 Motor controller

With the gearbox ratio changed, to keep the original tuner acceleration rates, the drive motor will have to accelerate four times slower. Software in the motor controller will have to be modified to maintain the same tuner acceleration and deceleration rates. The spare capacity left in the motor controller software is adequate to accommodate the proposed software changes for both continuous rotation and stepping modes.

6.3 Drive motor size

With tuner acceleration and deceleration rates remaining the same, the drive motor current is expected to remain at 40% of its full load current rating. The size of the drive motor was found be adequate to accommodate the insignificant increased in load due to wind resistance.

7. References

1 Gates, PowerGrip HTD Design Manual

2 Weeks, F., *Stress Analyses of an Energy Tuner used in a Electromagnetic Reverberation Chamber*, DSTO-TN-0272 May 2000

3 O'Higgins, P. J., *Basic Instrumentation Measurement*, first edition, 1966

Appendix 1
Structural Integrity

If tuner acceleration and deceleration rates are to be kept the same as those used when operating at 1rpm, then the original calculations for inertia loads, structural stresses, and safety factors are still valid [2]. The only additional loading will be the increased wind loading due to the higher rotational speed.

Wind loading at 4rpm

From page 181 reference [3] O'Higgins Basic Instrumentation Measurement.
Note that the formula and units used in the above reference are Imperial.

$h = v^2 \div 2g$

where
- h = head in pitot tube, ft
- v = fluid velocity, ft/s
- g = gravity 32.2ft/s^2

The formula converted from Imperial to SI units reads the same.

$h = v^2 \div 2g$

where
- h = head in pitot tube, metres
- v = fluid velocity, m/s
- g = gravity 9.81m/s^2

From page 65 [3]
The formula and units used in the above reference are again Imperial.

$p = h \times w_m$
$h = p \div w_m$

where
- p = pressure, lbs/ft^2
- h = head in pitot tube, ft
- w_m = weight density, lbs/ft^3
- w_m (air) = 0.075lbs/ft^3, page 186 [3], converted to SI units = 1.2014Kg/m^3

The formula converted from Imperial to SI units reads.

$h = p \div (w_m x\ g)$

where

p = pressure, N/m^2
h = head in pitot tube, m
w_m = weight density, Kg/m^3
g = 9.81 m/s^2

replace h in first formula ($h = v^2 \div 2g$) with $p \div (w_m \times g)$

$p \div (w_m \times g) = v^2 \div 2g$
$p = v^2 \div (2g) \times (w_m \times g)$
$p = v^2 \times w_m \div 2$

area of each paddle (3.6m x 3.6m) = $12.96m^2$

For the purpose of the calculations, consider the radial speed at the centre of the paddle to be the average linear velocity of the paddle.

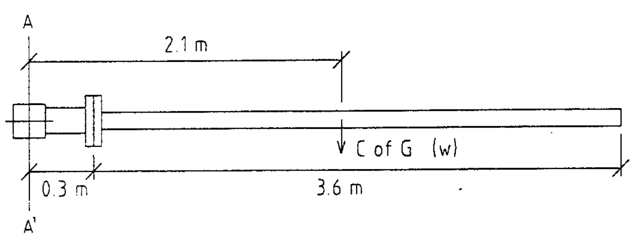

Figure 1-1 Paddle moment arm on main shaft

Distance travelled in one minute

= 4rpm x circumference at 2.1m radius = $4 \times 2 \times \pi \times 2.1m/min$
= 52.8m/min
= 0.88m/s

Air pressure on paddles

formula from above

$p = v^2 \times w_m \div 2$
$= 0.88^2 \times 1.2014 \div 2$
$= 0.4652N/m^2$

therefore

force on paddle= pressure x area

= 0.4652 x 12.96

= 6.03N

load on 4 paddles = 6.03x 4 = 24.1N

increased torque due to wind load = 24.1 x 2.1

= 50.6Nm

torque produced by one paddle = 12.7Nm

During continuous rotation the side load welds at the paddle root hub are only wind loads.

Stress in the vertical welds are

Data from Appendix 1
Weld data from [2]

Yield strength of weld material = 455 MPa
root thickness of weld = 5.6mm
length of weld (with 150mm strengthening webs)= 125 + 150 = 275mm
number of welds = 2
distance between welds= 0.125m

total area of side welds= 2 x 275x 5.6 = 3080mm^2

torque load on paddle welds= 12.7Nm

therefore load on welds = 12.7 ÷ 0.125

= 101.6N

stress in weld = load ÷ area

= 101.6 ÷ 3080

= 0.033MPa

safety factor = yield stress ÷ actual stress

= 455 ÷ 0.033

= 13,790

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19. ABSTRACT

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